



Proposed improvements to a model for characterizing the electrical and thermal energy performance of Stirling engine micro-cogeneration devices based upon experimental observations

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ABSTRACT

Stirling engines (SE) are a market-ready technology suitable for residential cogeneration of heat and electricity to alleviate the increasing demand on central power grids. Advantages of this external combustion engine include high cogeneration efficiency, fuel flexibility, low noise and vibration, and low emissions. To explore and assess the feasibility of using SE based cogeneration systems in the residential sector, there is a need for an accurate and practical simulation model that can be used to conduct sensitivity and what-if analyses. A simulation model for SE based residential scale micro-cogeneration systems was recently developed; however the model is impractical due to its functional form and data requirements. Furthermore, the available experimental data lack adequate diversity to assess the model's suitability. In this paper, first the existing model is briefly presented, followed by a review of the design and implementation of a series of experiments conducted to study the performance and behaviour of the SE system and to develop extensive, and hitherto unavailable, operational data. The empirical observations are contrasted with the functional form of the existing simulation model, and improvements to the structure of the model are proposed based upon these observations.

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1. Introduction

By generating heat and electricity through a single thermodynamic cycle, micro-cogeneration systems show promise to reduce energy costs and greenhouse gas emissions when compared with central generating plants [1–3]. One technology suitable for micro-cogeneration is the Stirling engine [1,3–5]. The Stirling engine utilizes heat from an external combustion process to drive pistons within sealed cylinders. A working fluid expands and contracts within each cylinder by means of a hot and a cold heat exchanger, forcing the piston to move in and out. A transmission system converts the linear piston motion to rotation that drives an alternator. Heat is provided to the hot end of the cylinder by an external flame, and cooling water removes heat from the cold cylinder end. The cooling water through the cold side of the piston in series with an exhaust gas heat exchanger provides the thermal output which can be used to supply domestic hot water, space heating, or (through a thermally activated cycle) space cooling. More detailed

descriptions of the Stirling cycle, its configurations and applications, can be found in [6,7].

The Stirling cycle has the highest theoretical efficiency of all gas power cycles [8]. In 2001, real Stirling cycle electrical efficiency was anticipated to reach 35–50%¹ at full load with an overall efficiency of 65–85% and at power-to-heat ratios of 1.2–1.7 [2]. However, more current literature indicates that modern SE micro-cogeneration devices have power-to-heat ratios of about 1:10, electrical efficiencies of 5–10%, and that overall efficiencies of field trials have achieved 77% [9,10]. Recent laboratory testing of one SE micro-cogeneration device using helium as its working fluid achieved electrical efficiencies between 13.8% and 23.6%, depending on the helium pressure [11]. It is believed that the variation in efficiencies is related to the power-to-heat ratio of the engine. In field trials, the power-to-heat ratio of micro-cogeneration technologies was found to be a major factor in both economic benefit [12,3] and carbon savings [9]. In these studies, SE micro-cogeneration devices with power-to-heat ratios of 1:10 were found to be more economic and carbon beneficial for the higher heat demands (compared to the demand for electricity) of most homes. However,

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¹ Efficiencies are reported in this article relative to the fuel's lower heating value, LHV.

it should be stressed that since the economic feasibility of micro-cogeneration devices is a strong function of the power-to-heat ratio as well as the feed-in tariff structure in effect, these observations on power-to-heat ratios are highly sensitive to feed-in tariff structures.

These substantial variations in performance are indicative of the early developmental stage of this technology. Even the core engine technology has not settled across manufacturers, and surely the optimal power-to-heat ratio for a marketable design is still heavily under examination. The technical and economic feasibility of a SE based micro-cogeneration system is a function of the design and size of the system as well as the building it is intended for. For this reason, an accurate and practical simulation model of SE micro-cogeneration devices is needed as a techno-economic analysis tool for studying and evaluating the performance of different SE micro-cogeneration devices in different load environments.

In the past it has been common to model the performance of SE micro-cogeneration devices using performance-map methods wherein the device's electrical and thermal efficiencies are treated as constant or are a parametric function of the device's loading [13–15]. This approach essentially decouples the modelling of the SE micro-cogeneration device from that of the house and other aspects of the thermodynamic system and as such precludes an accurate treatment of the thermal coupling to the building and its HVAC system, as demonstrated by [16]. In response to this shortcoming, a simulation model that characterizes the thermal and energy performance of combustion-based micro-cogeneration devices (including those based on SE) was developed within Annex 42 of the International Energy Agency's Energy Conservation in Buildings and Community Systems Programme (IEA/ECBCS), and is herein referred to as the Annex 42 model. The specification of this model is detailed in [17,18]. The Annex 42 model is one of the few combustion engine models suitable for use in whole-building simulation tools because of its parametric design and empirical nature. It has been calibrated to represent the performance of one particular SE micro-cogeneration device [11,18]. However, the data set used in the calibration was not ideal for this purpose as the data were gathered during typical operating cycles rather than at steady-state conditions.

The Annex 42 combustion cogeneration model has been used to assess the performance of both prototype and hypothetical SE micro-cogeneration devices in various climates [16,19]. These studies demonstrate the importance of accurately modelling start-up and shut-down behaviour to the overall estimation of system performance.

The present work describes a new experimental data set that was collected with the aim of critiquing, calibrating, and validating the Annex 42 model. Experiments were conducted on a prototype micro-cogeneration SE device to generate a data set comprising of both steady-state and transient modes of operation. Specific improvements to the Annex 42 model are proposed here to improve the treatment of steady-state, start-up, and shut-down sequences of operation.

2. Model description

This section outlines pertinent aspects of the Annex 42 model. The interested reader is referred to [17,18] for a detailed treatment of the model.

The Annex 42 model is a zero-order model. Unlike the first-, second-, and third-order models, as classified by [7], a zero-order model does not attempt to characterize thermodynamic behaviour. Rather, it can operate at a coarser time-step and can offer an appropriate level of precision within the context of whole-building simulation without requiring intrusive measurements for calibration.

The Annex 42 model represents both SE and ICE micro-cogeneration devices with three control volumes:

- the *energy conversion control volume* represents the engine working fluid, combustion gases, and engine alternator;
- the *engine thermal mass control volume* represents the aggregate thermal mass of the engine block and most internal heat exchange equipment; and
- the *cooling water control volume* represents the aggregate thermal mass of the cooling water and that portion of the heat exchanger in immediate thermal contact with the cooling fluid.

The energy and mass flows about these defined control volumes are depicted in Fig. 1.

The model employs an abstract representation wherein the energy conversion control volume represents the conversion of the fuel's chemical energy to electricity and heat under steady conditions. The dynamic thermal response of the system is then characterized by both the engine thermal mass and the cooling water control volumes. Given this, the energy balance of the energy conversion control volume is not formed or solved. Rather, its net electrical output under steady conditions ($P_{net,ss}$) and the thermal energy it transfers to the engine thermal mass control volume under steady conditions ($q_{gen,ss}$) are determined using overall conversion efficiencies that aggregate the effects of inefficiencies and auxiliary systems. Thus:

$$P_{net,ss} = \eta_e \dot{m}_{fuel} LHV_{fuel} \quad (1.a)$$

$$q_{gen,ss} = \eta_q \dot{m}_{fuel} LHV_{fuel} \quad (1.b)$$

where $\eta_e (-)$ and $\eta_q (-)$ are the steady-state part load electrical and thermal efficiencies, respectively, $\dot{m}_{fuel}$ (kg s^{-1} or kmol s^{-1}) is the fuel flow rate, and LHV_{fuel} (J kg^{-1} or J kmol^{-1}) is the lower heating value of the fuel.

The following energy balance characterizes the system,

$$q_{gen,ss} = (\dot{m}c_p)_{cw}(T_{cw,out} - T_{cw,in}) + U_{loss}(T_{eng} - T_{room}) + (Mc_p)_{eng} \times \frac{dT_{eng}}{dt} + (Mc_p)_{HX} \frac{dT_{cw,out}}{dt} \quad (2)$$

where $U_{loss} (-)$ is the coefficient of heat loss for the aggregate engine mass, T_{eng} ($^{\circ}\text{C}$) and T_{room} ($^{\circ}\text{C}$) represent the average engine

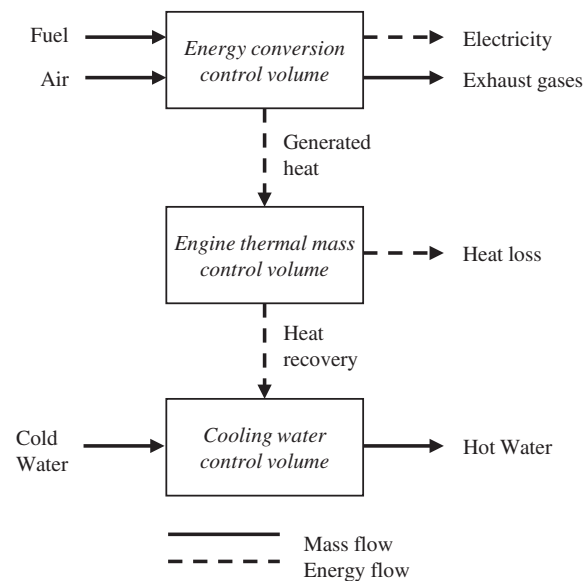


Fig. 1. Discretization scheme employed by Annex 42 model (adapted from Ferguson et al. [18]).

and room temperatures, respectively, $(\dot{m}c_p)_{cw}$ is the product of flow rate and heat capacity of the cooling water, $(Mc_p)_{eng}$ and $(Mc_p)_{HX}$ are the engine and heat exchanger control volume heat capacities, respectively, and $T_{cw,in}$ (°C) and $T_{cw,out}$ (°C) are the temperatures of the cooling water entering and exiting the engine control volume, respectively.

The final two terms in Eq. (2) represent the transient energy storage of the engine thermal mass and cooling water control volumes. Under steady-state conditions, the energy balance in the engine thermal mass control volume reduces to:

$$q_{gen,ss} = (\dot{m}c_p)_{cw}(T_{cw,out} - T_{cw,in}) + U_{loss}(T_{eng} - T_{room}) \quad (3)$$

where the first term on the right side of the equation represents the device's useful thermal output.

In the Annex 42 model, the steady-state conversion efficiencies are represented as truncated tri-variate polynomial functions of cooling water mass flow rate ($\dot{m}_{cw}$), cooling water inlet temperature ($T_{cw,in}$), and net electrical power output at steady-state ($P_{net,ss}$). These expressions are given in Eqs. (4) and (5), respectively:

$$\begin{aligned} \eta_e = & a_0 + a_1 P_{net,ss}^2 + a_2 P_{net,ss} + a_3 \dot{m}_{cw}^2 + a_4 \dot{m}_{cw} + a_5 T_{cw,in}^2 \\ & + a_6 T_{cw,in} + a_7 P_{net,ss}^2 \dot{m}_{cw}^2 + a_8 P_{net,ss} \dot{m}_{cw} + a_9 P_{net,ss} \dot{m}_{cw}^2 \\ & + a_{10} P_{net,ss}^2 \dot{m}_{cw} + a_{11} P_{net,ss}^2 T_{cw,in}^2 + a_{12} P_{net,ss} T_{cw,in} \\ & + a_{13} P_{net,ss} T_{cw,in}^2 + a_{14} P_{net,ss}^2 T_{cw,in} + a_{15} \dot{m}_{cw}^2 T_{cw,in}^2 \\ & + a_{16} \dot{m}_{cw} T_{cw,in} + a_{17} \dot{m}_{cw} T_{cw,in}^2 + a_{18} \dot{m}_{cw}^2 T_{cw,in} \\ & + a_{19} P_{net,ss}^2 \dot{m}_{cw}^2 T_{cw,in}^2 + a_{20} P_{net,ss}^2 \dot{m}_{cw}^2 T_{cw,in} \\ & + a_{21} P_{net,ss}^2 \dot{m}_{cw} T_{cw,in}^2 + a_{22} P_{net,ss} \dot{m}_{cw}^2 T_{cw,in}^2 \\ & + a_{23} P_{net,ss}^2 \dot{m}_{cw} T_{cw,in} + a_{24} P_{net,ss} \dot{m}_{cw}^2 T_{cw,in} \\ & + a_{25} P_{net,ss} \dot{m}_{cw} T_{cw,in}^2 + a_{26} P_{net,ss} \dot{m}_{cw} T_{cw,in} \end{aligned} \quad (4)$$

$$\begin{aligned} \eta_q = & b_0 + b_1 P_{net,ss}^2 + b_2 P_{net,ss} + b_3 \dot{m}_{cw}^2 + b_4 \dot{m}_{cw} + b_5 T_{cw,in}^2 \\ & + b_6 T_{cw,in} + b_7 P_{net,ss}^2 \dot{m}_{cw}^2 + b_8 P_{net,ss} \dot{m}_{cw} + b_9 P_{net,ss} \dot{m}_{cw}^2 \\ & + b_{10} P_{net,ss}^2 \dot{m}_{cw} + b_{11} P_{net,ss}^2 T_{cw,in}^2 + b_{12} P_{net,ss} T_{cw,in} \\ & + b_{13} P_{net,ss} T_{cw,in}^2 + b_{14} P_{net,ss}^2 T_{cw,in} + b_{15} \dot{m}_{cw}^2 T_{cw,in}^2 \\ & + b_{16} \dot{m}_{cw} T_{cw,in} + b_{17} \dot{m}_{cw} T_{cw,in}^2 + b_{18} \dot{m}_{cw}^2 T_{cw,in} \\ & + b_{19} P_{net,ss}^2 \dot{m}_{cw}^2 T_{cw,in}^2 + b_{20} P_{net,ss}^2 \dot{m}_{cw}^2 T_{cw,in} \\ & + b_{21} P_{net,ss}^2 \dot{m}_{cw} T_{cw,in}^2 + b_{22} P_{net,ss} \dot{m}_{cw}^2 T_{cw,in}^2 \\ & + b_{23} P_{net,ss}^2 \dot{m}_{cw} T_{cw,in} + b_{24} P_{net,ss} \dot{m}_{cw}^2 T_{cw,in} \\ & + b_{25} P_{net,ss} \dot{m}_{cw} T_{cw,in}^2 + b_{26} P_{net,ss} \dot{m}_{cw} T_{cw,in} \end{aligned} \quad (5)$$

where a_0 to a_{26} and b_0 to b_{26} are empirically derived coefficients.

As can be seen, Eqs. (4) and (5) each require 27 empirical coefficients to characterize the conversion efficiencies. Available experimental test data to date have not permitted model calibration of all 54 regression coefficients (a_0 – a_{26} and b_0 – b_{26}). Indeed, [18] considered only the constant terms (a_0 and b_0) in their calibration. The appropriateness of the functional form of Eqs. (4) and (5) is examined in Section 5 using the experimental data described in Sections 3 and 4.

Combustion-based cogeneration systems ultimately cycle their operation in response to the load. The model description presented above characterizes normal operation, where the engine is on and operating steadily or modulating (by cycling on and off) in response to a dynamic load. The Annex 42 model's treatment of stand-by, start-up², and shut-down modes of operation is discussed next.

During stand-by operation, which is also an equilibrium state, the engine consumes a small amount of electricity to power a control board. The duration of stand-by mode has no limit; it starts when shut-down is complete and ends when the engine is required to start to meet a thermal or electrical load. There is no fuel consumption in stand-by, and thermal output is assumed to be zero. The only parameter used for calibration is the net electrical output, assumed constant (and invariably negative) in stand-by mode:

$$P_{net} = P_{net, stand-by} \quad (6.a)$$

$$Q_{gen, ss} = 0 \quad (6.b)$$

$$\dot{m}_{fuel} = 0 \quad (6.c)$$

After the start-up period is initiated by a call for either thermal or electrical production, the SE micro-cogeneration device exhibits behaviour where the fuel flow and electrical output differ from their steady-state values. Because some of the fuel energy added is used to heat the engine, the steady-state power can not immediately be achieved. To accelerate power production, the engine's fuel controller may request more fuel during start-up.

The start-up modes of SE micro-cogeneration devices are characterized as being affected by the engine's temperature which, during start-up, is less than nominal. The Annex 42 model represents equivalent steady-state conditions by a *nominal engine temperature* ($T_{eng,nom}$). In start-up mode, the engine controller is assumed to both adjust the engine fuel flow to the value corresponding to maximum power output, and to further increase this value according to the deviation of the actual engine temperature (T_{eng}) from the nominal value. Thus, the fuel flow at start-up is determined by³:

$$\dot{m}_{fuel, start-up} = \dot{m}_{fuel, ss-max} + k_f \dot{m}_{fuel, ss-max} \left(\frac{T_{eng, nom} - T_{room}}{T_{eng} - T_{room}} \right) \quad (7)$$

where $k_f(-)$ is an empirical coefficient and $\dot{m}_{fuel, ss-max}$ (kg s^{-1}) is the maximum achievable fuel flow at steady-state.

The net electricity produced during start-up ($P_{net, start-up}$) is similarly correlated to the nominal temperature:

$$P_{net, start-up} = P_{max} k_p \left(\frac{T_{eng} - T_{room}}{T_{eng, nom} - T_{room}} \right) \quad (8)$$

where $k_p(-)$ is an empirical coefficient and P_{max} (W) is the engine's maximum (net) steady-state electrical output.

The steady-state thermal efficiency is then evaluated and the heat output of the engine is determined using Eq. (1.b). From here, the thermal response of the engine and cooling water control volumes are addressed.

The SE micro-cogeneration device enters into normal operation whenever:

- (i) the engine temperature reaches the nominal value, or
- (ii) the net power produced exceeds that requested by the controller.

The Annex 42 model defines shut-down period as beginning when the fuel supply has stopped and ending when stand-by mode is entered. It includes the ramping down and shut-down of the alternator and the cool-down period when the cooling water pump and draft fan operate. The Annex 42 model assumes no fuel consumption during shut-down, and assumes the net electrical generation is a single, constant parameter. Thermal output during shut-down is non-zero since residual heat in the engine is transferred to the cooling water.

² Annex 42 combustion cogeneration model specification uses terminology "warm-up" and "cool-down" periods rather than start-up and shut-down.

³ Practical limitations of Eqs. (7) and (8) are addressed in [11].

The appropriateness of the treatment of the start-up and shut-down modes of operation is examined in Section 6 of the current paper using the experimental data described in Sections 3 and 4.

3. Experimental and measurement procedures

An experimental test matrix was designed to evaluate the steady-state performance of a prototype 750W_{DC} SE micro-cogeneration device. The test matrix was derived from the Annex 42 test protocol [11] with emphasis placed upon tests which were most relevant for calibrating and examining the validity of the Annex 42 model.

3.1. Experimental test matrix

The Annex 42 experimental protocol includes *steady-state* testing at a range of constant coolant inlet temperatures and flow rates, at 100% and reduced settings of electrical output. A thermal response test is recommended where coolant inlet temperature is ramped from 10 °C to 90 °C. *Dynamic* tests include operating the cogeneration device from a cold start to steady-state, and monitoring shut-down from steady-state until there is no significant temperature difference across the device's heat exchanger. Finally, *electrical and thermal load-following* tests are recommended where the device operates normally according to its own controls in a residential load setting.

The experimental test matrix for this work includes steady-state and dynamic testing. Load-following test were not conducted because the prototype device normally operates at full power only. In addition, power consumption data obtained during stand-by operation are not reported here because the magnitude was smaller than the calibration range of the power meter used. However, it was observed that the Annex 42 model's assumption of constant electrical consumption was valid for the tested SE device, and it was concluded that no modification of the Annex 42 model is required in this respect.

The range of engine coolant inlet temperatures that were achievable with the given test set-up was 25–70 °C. The range of coolant flow rates achievable was 3–9.5 L min⁻¹. As coolant temperature increased and flow decreased, the engine was more likely to overheat. This, combined with test program timing limitations, restricted the test matrix to the combinations of coolant temperature and flow given in Table 1.

The maximum engine coolant inlet temperature was limited to about 70 °C by the engine controllers' thermal protection circuit.

The experimental setup enabled a minimum engine coolant inlet temperature of 30 °C.

The SE manufacturer's minimum, maximum, and recommended coolant flow rates are 5, 8, and 6.5 slpm (standard litres per minute), respectively. The engine was operated beyond these recommended values for short periods of time under constant observation with no apparent ill effects. Since the Annex 42 protocol recommends data at 50% of the manufacturer's recommended coolant flow rate, a minimum of 3.5 slpm was used. The maximum coolant flow rate was 9 slpm; although this is less than the 200% of the manufacturer's recommended value required by the Annex 42 protocol, this upper limit was the maximum achievable with the pump.

The SE micro-cogeneration device tested can operate at a reduced power level only when all of the following three conditions exist:

- The engine is in a user-selectable mode to follow the thermal load.
- The batteries are completely charged, and
- A heat demand exists.

In this situation the device's control system reduces the power to the draft fan and the fuel regulator adjusts the fuel flow, resulting in lowered electrical (and thermal) output. Some electrical energy is then diverted through a clamp heating element to further heat the cooling water. The engine controls allow no more than four hours of operation in this mode. Since this is not a normal operating mode, no experiments were conducted at the reduced power output. An AC electrical load was drawn from the batteries through a sine wave inverter to prevent the batteries from being fully charged and avoid any use of the clamp heating element.

3.2. Experimental apparatus and instrumentation

A schematic diagram of the test apparatus detailing thermal and instrumentation components is shown in Fig. 2. A Campbell Scientific CR23X datalogger was used to record the bulk of the test measurements. The engine commissioning software, 'Micromon', was available for monitoring and logging internal engine parameters.

The quantities measured during the experiments and the instrumentation employed are provided in Table 2. Each quantity was measured and logged at 10-s intervals. The instrumentation bias errors associated with each measurement are also given in Table 2.

Table 1
Experimental test matrix summary.

	Coolant flow rate (L min ⁻¹)	Coolant inlet temp. (°C)	Description
Steady-state ~30 min (SS1–SS27)	3.5	30, 50	Minimum achievable flow rate
	5.0	30, 50, 70	Minimum recommended flow rate
	6.5	30, 50, 70	Actual recommended flow rate
	8.0	30, 50, 70	Maximum recommended flow rate
	9.5	30, 50, 70	Maximum achievable flow rate
Transient start-up (SU1–SU7)	6.5	25–70	Cold start, temperature ramp up
	Varied	Varied	Cold start
	Varied	Varied	Hot start
Transient shut-down (SD1–SD11)	Varied	Varied	Full power to off
	6.5	65	Full power ramp down to 50% over 20 min
Stand-by (SB1–SB6)	N/A	N/A	Stand-by mode
Emissions test	Varied	Varied	Monitoring flue gas during start-up, normal operation, shut-down
Start–stops	6.5	~35	5 repetitions of start-stop in 1 h 15 m
Winter battery manage mode	6.5	Auto	75% of original defined electrical load was applied to avoid depleting batteries; 24-h test

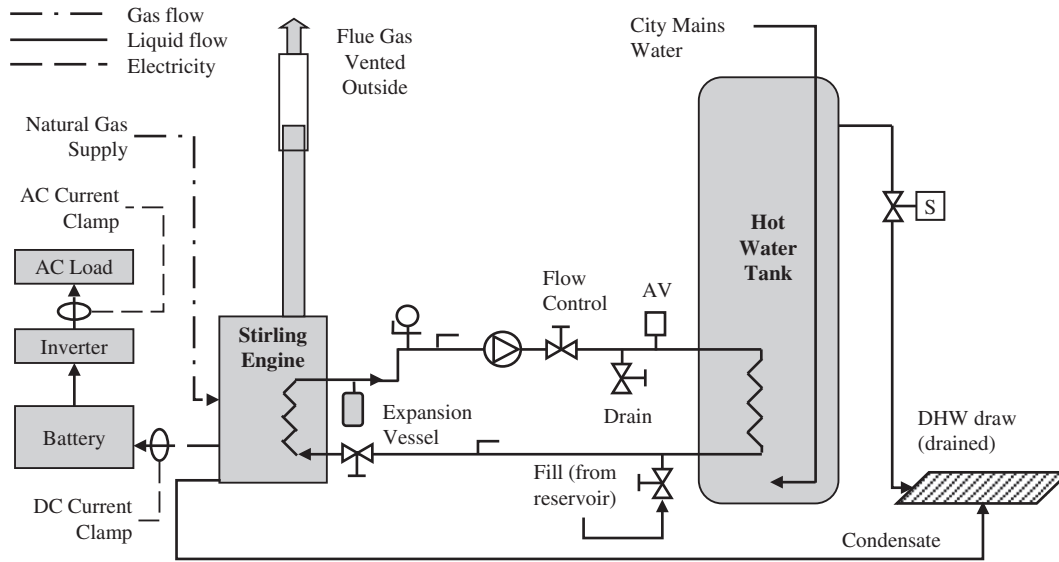


Fig. 2. Experimental apparatus to achieve steady-state conditions.

To increase the accuracy of the flow control and heat balance calculations, a rake of six thermocouples was constructed to measure a temperature profile within the hot water tank (HWT). Thermocouples were spaced throughout the tank and an average temperature was used to represent the internal temperature of the tank.

The coolant flow rate was controlled by a manual valve (see “flow control” valve in Fig. 2) and monitored in real-time. The flow rate fluctuated moderately with both engine power, because of varying bus voltage, and with coolant temperature, relating to fluid pressure. Fluctuations in flow rate combined with the pulsed measurement made the flow rate difficult to control, particularly during transient tests such as start-up and shut-down.

The SE micro-cogeneration device's commissioning software, Micromon, recorded intrusive measurements including gross shaft power, the temperature of exhaust gas entering the exhaust gas heat exchanger (EGHX), and the air supply rate. The intrusive proprietary data are not directly reported herein, but have been used to derive other quantities such as parasitic losses within the SE micro-cogeneration device.

When the coolant inlet temperature is below a certain threshold, water vapour in the exhaust gases condenses in the engine's exhaust gas heat exchanger. Condensate from the SE micro-cogeneration device was normally drained through a hose. However, during one set of tests with varying coolant inlet temperature, condensate was collected in a beaker in order to estimate the coolant inlet temperature threshold at which condensation begins. It also provided an estimate of condensate flow rate at low coolant inlet temperatures.

A steady-state condition was imposed on the engine coolant inlet temperature by comparing the temperature to the desired value in a conditional statement and draining water from the hot water tank based on this comparison. The CR23X datalogger was used to both evaluate the condition, and to supply a control signal to a solenoid in the tank drainpipe to drain the hot water. With cold water replenishing the drained hot water, the tank temperature moderated the coolant temperature circulating through its coils and through the engine. Some vacillation in coolant temperature was observed due to the system dynamics and attempts to minimize the magnitude of vacillation resulted in slightly different conditional statements for low and high temperature tests.

The version of the SE micro-cogeneration device that was tested has a direct current output of 24 volts that is used to charge a bat-

tery bank. The power sent to the batteries is the net DC electrical output after parasitic losses from the unit's auxiliaries.

The battery bank used for the experiment was a set of six deep-cycle gel cell lead–acid batteries, each with a capacity of 75 Ah and a voltage of 12 V DC. The batteries were connected with two in series and three in parallel to create a battery bank with a capacity of 225 Ah and a voltage of 24 V DC.

4. Calculation of derived quantities and propagation of measurement uncertainties

This section presents definitions of thermal and electrical efficiency that can be calculated from the measured data specified in Table 2.

A common definition for thermal efficiency is the ratio of heat extracted by the engine coolant to the energy content of the fuel:

$$\eta_q = \frac{Q_{HX}}{\dot{m}_{fuel} LHV_{fuel}} = \frac{\dot{m}_{cw} c_{cw} \Delta T_{cw}}{\dot{m}_{fuel} LHV_{fuel}} \quad (9)$$

where c_{cw} is the specific heat capacity of the cooling water, LHV_{fuel} is the lower heating value of the fuel (natural gas), and ΔT_{cw} is the temperature rise of the cooling water across the engine, including all heat exchange to the cooling water within the engine.

The coolant circuit within the device tested extracts heat from the generator, the (four) cylinder heat exchangers, and the exhaust gas heat exchanger, in addition to the general heat gain within the engine enclosure. Although the magnitude of each of these components are not quantifiable from the measured data described in Section 3, the overall temperature rise of the coolant water across the engine is sufficient to determine the thermal efficiency as defined in Eq. (9). Given this, the thermal efficiency can be determined directly from the data measured during the steady-state tests outlined in Section 3.

However, Eq. (9) differs from the representation employed in the Annex 42 model, as can be seen by inspecting Eqs. (1) and (3). In the Annex 42 model, the thermal efficiency cannot be determined directly from measured data taken during steady-state experiments due to the U_{loss} term. Upon review, the definition for thermal efficiency implicit in Eq. (3) is not consistent with the definition in Eq. (9). Since the definition in Eq. (9) is a more common representation of thermal efficiency and because it facilitates calibration of the thermal efficiency using data gathered during stea-

Table 2
Measured quantities and instrumentation.

Parameter	Symbol	Units	Datalogger	Instrument	Range (min–max)	Instrumentation bias error ^h tol±
Gross DC production prior to power conditioning	$P_{gross,DC}$	W	Micromon ¹	Unknown	Unknown	1% of reading assumed
Net DC electrical output to battery storage	$P_{net,DC}$	W	DC PowerSight	Summit ^a PS250 & DC600 clamp	5–600 A; 600 V maximum	1 A or 2% of reading
Net AC output after parasitic, battery, and PCU losses	$P_{net,AC}$	W	AC PowerSight/CR23X ^b	Summit PS250 A3 Elster ^c	0.1–100 A; 600 V maximum	0.2 A or 2% of reading 0.2% of reading
Natural gas flow rate	$\dot{m}_{fuel}$	slpm	CR23X	Sierra ^d 822S Top-Trak TM	0–50	1% FS ⁱ
Air supply rate	$\dot{m}_{air}$	% of max	Micromon	Unknown	0–120	Unknown
Ambient air temperature	T_{amb}	°C	CR23X	Omega ^e Type T Thermocouple	0–350	1 °C or 0.75% of reading
Volume flow rate of coolant to SE	v_{cw}	Lmin ⁻¹	CR23X	Omega FTB4607 (Dec. 3–4, 2007)	0.6–75	1.5% of reading
Temperature of exhaust gas entering EGHX	T_{exh}	°C	Micromon	Omega Type K thermocouple	0–1250 for given <i>tol</i>	2.2 °C or 0.75% of reading
Temperature of exhaust gas exiting EGHX	T_{flue}	°C	CR23X	Omega Type T thermocouple	0–350 for given <i>tol</i>	1 °C or 0.75% of reading
Temperature of coolant exiting engine	$T_{cw,out}$	°C	CR23X	Omega Type T thermocouple	0–350 for given <i>tol</i>	1 °C or 0.75% of reading; calibrated to ±0.5 °C
Temperature of coolant entering engine	$T_{cw,in}$	°C	CR23X	Omega Type T thermocouple	0–350 for given <i>tol</i>	1 °C or 0.75% of reading; calibrated to ±0.5 °C
Volume flow rate of cold water makeup to tank	$v_{ColdWat}$	Ls ⁻¹	CR23X	GWLF ^f MTH high temperature 1 pulse per litre	2 minimum	3% of reading
Temperature of water in storage tank	$T_{HWT,1} - T_{HWT,6}$	°C	CR23X	Omega Type T Thermocouple	0–350 for given <i>tol</i>	1 °C or 0.75% of reading
Temperature of DHW	T_{DHW}	°C	CR23X	Omega Type T Thermocouple	0–350 for given <i>tol</i>	1 °C or 0.75% of reading
Temperature of City Water	$T_{ColdWat}$	°C	CR23X	Omega Type T Thermocouple	0–350 for given <i>tol</i>	1 °C or 0.75% of reading
Exhaust gas composition (molar fractions of CO ₂ , NO _x , O ₂ , CO)	$M_{CO_2}, M_{NO_x}, M_{O_2}, M_{CO}$	mV (%) or ppm)	CR23X	Horiba ^g PG250	Variable; note 0–5000 mV output is linearly proportional to measurement scale range selected on Horiba	1% of reading assumed (unit was calibrated every 2–3 h, zero drift error of 1% FS ⁱ per day)
Engine chassis surface temperatures	$T_{LHS}, T_{RHS}, T_{front}, T_{back}, T_{top}$	°C	CR23X	Omega Type K surface thermocouple	0–1250 for given <i>tol</i>	2.2 °C or 0.75% of reading

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^f GWF MessSysteme AG, Obergundstrasse 119, CH-6002, Switzerland. Tel.: +41 41 310 60 87; www.gwf.ch.

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^h Whichever value is greater.

ⁱ FS: full scale.

dy-state experiments, it is recommended that the U_{loss} term in Eqs. (2) and (3) be removed from the engine thermal mass control volume heat balance. The resulting expression in Eq. (10) includes energy loss from the engine control volume to the room within the thermal efficiency term:

$$q_{gen,ss} = \eta_q \dot{m}_{fuel} LHV_{fuel} = (\dot{m}_p)_{cw} (T_{cw,out} - T_{cw,in}) \quad (10)$$

The impact that conglomerating the U_{loss} term into the thermal efficiency term has upon the transient response of the model will be assessed in a subsequent paper that will explore the validity of this form using measured data.

The electrical efficiency is defined as the ratio of net direct current power output from the engine to the gross energy content of the fuel:

$$\eta_e = \frac{P_{net,DC}}{\dot{m}_{fuel} LHV_{fuel}} \quad (11)$$

where $P_{net,DC}$ is the net power production of the engine.

The gross power was known by an internal measurement reported by the engine commissioning software, while the net power production was measured. Thus, the parasitic loss of the engine can be determined from these two quantities:

$$P_{parasitic} = P_{gross,DC} - P_{net,DC} \quad (12)$$

The parasitic losses include consumption for the draft fan to draw air through the combustion system, the coolant circulation pump, control board power consumption, and a small amount of transmission losses.

To understand the significance of experimental test results, an uncertainty analysis must be carried out with the calculations. From known or estimated instrument bias error, a band of certainty for each calculated result can be identified.

Uncertainties of measured and derived quantities were calculated using the method recommended by the American Society of Mechanical Engineers [20,21]. A bias error is assigned to each measured quantity that results from the instrumentation

specifications, and sometimes includes a reading error or other error components specific to the measurement.

The total bias for each measured parameter, p , at a recorded data interval, i , is calculated as the sum of the squares of bias error components for that measurement:

$$B_{pi} = \sqrt{B_1^2 + B_2^2 + \dots + B_k^2} \quad (13)$$

where $B_1, B_2, \dots, B_k$ are the bias values from ' k ' components of error for measurement p . The measurement instrument's quoted accuracy was used as the bias error when known. Any unknown bias errors of intrusive measurements provided by the SE micro-cogeneration device's commissioning software were assumed to be 1% of the reading.

The precision index, S , of each recorded data interval was calculated as the standard deviation of the data set:

$$S_p = \sqrt{\frac{\sum_{i=1}^N (x_i - \bar{x})^2}{N - 1}} \quad (14)$$

where N is the number of recorded data intervals in the set, x_i is the measured data at each interval, i , and $\bar{x}$ is the mean value for the set. By this definition, the precision index is constant across a data set. The precision error for a non-steady data set can be high and misleading since the mean is not of particular statistical importance and neither, therefore, is the deviation from the mean.

If the average value from a set of either a measured or a derived quantity is to be reported, the precision index of the average value, S_{avg} is given by Eq. (15):

$$S_{avg} = \frac{S_p}{\sqrt{N}} \quad (15)$$

The total uncertainty of a measured quantity is calculated by combining the bias and precision errors:

$$U_{95\%} = \sqrt{B_{pi}^2 + (tS_p)^2} \quad (16.a)$$

$$U_{99\%} = B_{pi} + tS_p \quad (16.b)$$

where $U_{95\%}$ and $U_{99\%}$ are the uncertainties for 95% and 99% two-sided confidence levels, respectively, and t is the Student's t value, evaluated as a function of N [22]. When the uncertainty of an average value is calculated, the S_{avg} value from Eq. (15) is used in Eqs. (16a and b). The uncertainty of a derived quantity is propagated via the bias and precision indices of measured quantities presented in Eqs. (13) and (14). The final bias error of a calculated value, v , is determined using sensitivity factors, θ , which propagate the error in measured parameters to the error in the final value. The sensitivity factors are defined as the partial derivatives:

$$\theta_i = \frac{\partial v}{\partial p_i} \quad (17)$$

The bias in the calculated value is then determined by:

$$B_v = \sqrt{\sum_{i=1}^j (\theta_i B_{pi})^2} \quad (18)$$

where the derived value, v , is a function of j measured parameters $p_1, p_2, \dots, p_j$, and $B_{p,i}$ is the bias of parameter p at interval i , calculated in Eq. (13) for data index i .

A similar equation applies for the precision index of the calculated value, S_v , using $S_{p,i}$ from Eq. (14). The total uncertainty is then calculated by Eqs. (16a and b).

For example, the uncertainty in the average value of the thermal efficiency (Eq. (9)) for a set of N samples is derived by first applying Eq. (13) to determine the bias error of η_q from those of its components:

Table 3

Uncertainty parameters for test SS-1 at 30 °C and 9.5 L min⁻¹.

Measurement	Average value	B (±)	S_{avg} (±)	$U_{95\%}$ (±)
$T_{cw,in}$ (°C)	30.18	0.5	0.013	0.5
$T_{cw,out}$ (°C)	40.16	0.5	0.0051	0.5
$\dot{m}_{fuel}$ (L min ⁻¹)	13.00	1.05	0.01	1.1
$P_{net,DC}$ (W)	812.31	16.25	0.44	16
$\dot{m}_{cw}$ (L min ⁻¹)	9.706	0.042	0.0019	0.042

$$\eta_q = \frac{Q_{HX}}{\dot{m}_{fuel} LHV_{fuel}} = \frac{\dot{m}_{cw} c_{cw} \Delta T_{cw}}{\dot{m}_{fuel} LHV_{fuel}} \quad (19)$$

$$B_{\eta_q} = \sqrt{\left(\frac{\partial \eta_q}{\partial Q_{HX}} * B_{Q_{HX}}\right)^2 + \left(\frac{\partial \eta_q}{\partial \dot{m}_{fuel}} * B_{\dot{m}_{fuel}}\right)^2 + \left(\frac{\partial \eta_q}{\partial LHV_{fuel}} * B_{LHV_{fuel}}\right)^2} \quad (20)$$

and $B_{Q_{HX}}$ is calculated similarly from its component biases. The precision index for each value in the set is similarly calculated:

$$S_{\eta_q} = \sqrt{\left(\frac{\partial \eta_q}{\partial Q_{HX}} * S_{Q_{HX}}\right)^2 + \left(\frac{\partial \eta_q}{\partial \dot{m}_{fuel}} * S_{\dot{m}_{fuel}}\right)^2 + \left(\frac{\partial \eta_q}{\partial LHV_{fuel}} * S_{LHV_{fuel}}\right)^2} \quad (21)$$

The precision of the average value is then used to calculate the total uncertainty in the derived mean thermal efficiency with 95% confidence:

$$S_{avg, \eta_q} = \frac{S_{\eta_q}}{\sqrt{N}} \quad (22)$$

$$U_{\eta_q, 95\%} = \sqrt{B_{\eta_q}^2 + (tS_{avg, \eta_q})^2} \quad (23)$$

For a sample steady-state test, Table 3 gives bias and precision indices of the five raw measurements that contribute to the uncertainty in electrical, thermal, and total efficiency. All uncertainties reported herein are with respect to a 95% confidence interval.

Although the T-type thermocouples used to measure cooling water temperatures have raw bias errors of ± 1 °C each, the pair measured an average difference of 0.015 °C in a 25 °C water bath. This is an indication that the two thermocouples yield a measurement of the temperature difference more accurate than the manufacturer's listed instrument bias of each thermocouple. Since only the temperature difference is used in Eq. (9), the manufacturer's special limit of error value of ± 0.5 °C was applied as the bias error of each thermocouple for a more realistic estimation of measurement uncertainty.

5. Proposed improvements to modelling electrical and thermal conversion efficiencies

One limitation of the Annex 42 model's treatment of steady-state efficiency is that each 27-term polynomial requires at least that many data points for its calibration. Another problem is that the coefficients have no clear physical significance: any values for net electrical output, coolant temperature and flow rate will perturb the constant term. Finally, a well-known characteristic of higher-order polynomials is that they tend to be very ill-conditioned; in this case the 27-term form is not only sensitive to measurement uncertainty, but it can behave poorly (with high oscillation amplitudes) between calibration points. A simplified steady-state performance characteristic could be more practical to calibrate and yield a more accurate result overall.

Based upon an examination of the measured data, an alternate functional form has been inferred to relate the steady-state efficiencies to three predictor variables: coolant flow rate, coolant

inlet temperature, and net electrical output. The proposed alternate forms to Eqs. (4) and (5) are given as follows:

$$\eta_e = \alpha_0 + \alpha_1(P_{net,ss} - P_{net,ss,REF}) + \alpha_2(T_{cw,in} - T_{cw,in,REF}) + \alpha_3(\dot{m}_{cw} - \dot{m}_{cw,REF}) \quad (24)$$

$$\eta_q = \beta_0 + \beta_1(P_{net,ss} - P_{net,ss,REF}) + \beta_2(T_{cw,in} - T_{cw,in,REF}) + \beta_3(\dot{m}_{cw} - \dot{m}_{cw,REF}) \quad (25)$$

This linear difference model is still a zero-order model relying heavily on empirical data. However, at nominal operating conditions, the efficiency is equal to the constant term. This makes the nominal efficiency a likely output of the proposed linear difference model, with small perturbations away from the nominal value.

Based upon an examination of the experimental data set and following attempts to calibrate the linear difference model with various reference values, the most appropriate reference values were determined. These were found to be the maximum observed net power output of the engine ($P_{net,ss,REF}$), the coolant inlet temperature at full power at which condensation is steadily produced ($T_{cw,in,REF}$), and the manufacturer's recommended coolant flow rate ($\dot{m}_{cw,REF}$).

Statistical methods were used to evaluate potential alternate model forms. The regression coefficients, p -values for each coefficient, output residuals, and coefficient of determination are all described here. The discussion of the statistical tests for these possible model replacements given below show that the linear difference model (Eqs. (24) and (25)) is the most practical replacement for the steady-state electrical and thermal conversion efficiency equations in the Annex 42 model (Eqs. (4) and (5)).

The first indication of the suitability of a proposed model is to examine the magnitude and sign of each regression coefficient. For example, the sign of β_2 in Eq. (25) should be known based on the choice of $T_{cw,in,REF}$, and an understanding from thermodynamics that the thermal efficiency of a heat engine increases with the temperature difference between the hot and cold cylinder ends. Coefficients large in magnitude (in the order of 10^4) could result in the highly oscillatory behaviour mentioned earlier, whereas small coefficients (in the order of 10^{-8}) could identify terms not significant to the result.

As a measure of statistical significance of each term in the steady-state efficiency characteristics, the p -value can be used. In the context of calibrating parametric equations, the p -value represents the probability that, if a term is removed from the equation, a result at least as extreme will occur given that the coefficient is zero [23]. Statistical significance does not always imply practical significance, and the sample size affects the p -value, so some discretion is required.

The p -value is calculated for each term by setting that coefficient to zero while keeping the others non-zero and evaluating the new polynomial. Comparison of the p -value is made with a chosen level of significance to determine each term's statistical significance:

$$\text{For statistical significance: } P(\eta \geq \eta_{observed}) \leq \alpha \quad (26)$$

where P is the probability, η is the fitted result with one coefficient set to zero, $\eta_{observed}$ is the fitted result with all coefficients non-zero, $\alpha = 100 - V/100$, the chosen significance level, and V (%) is the double-sided confidence interval.

For a confidence interval of 95%, the value of α is 0.05. However, even if none of the terms are statistically significant at the chosen α -level, the conclusion cannot be made, solely on this result, that all of the terms are statistically insignificant. It can be the case that the p -values indicate statistical insignificance for terms at a chosen value of α , but that together the terms are significant [23]. This is more likely to occur as the number of terms increases, given that each term then holds less weight in the result.

Another common test to assess the suitability of the proposed model is to look for trends in the output residual [23]. The output residual is the difference between the expected result (calculated from measured data) and the fitted result (determined from data regression). In the steady-state efficiency characteristics, the output is either thermal or electrical efficiency, and the residual is simply calculated by:

$$\text{Residual} = \eta_{expected} - \eta_{fitted} \quad (27)$$

The output residual is plotted against both the output and predictor variables to identify if a correlation was missed in the regression. If a trend in a set of residuals is visible, it is possible that one or more terms are missing in the correlation.

Finally, the coefficient of determination (r^2) is a measure of how well future measurements are likely to be predicted by the model. It is calculated as one minus the ratio of the error sum of squares to the total sum of squares:

$$r^2 = 1 - \frac{\sum_{i=1}^N (\eta_{i,expected} - \eta_{i,fitted})^2}{\sum_{i=1}^N (\eta_{i,expected} - \eta_{i,average})^2} \quad (28)$$

The coefficient of determination cannot be the only method used to evaluate a proposed model, since it would reward the highest order, and likely most ill-conditioned, polynomial.

The regression coefficients α_0 – α_3 and β_0 – β_3 and p -values for each coefficient of the linear difference model are given in Table 4. Regression coefficients were calibrated using a multiple linear regression function in MATLAB [22], which employs a least squares method.

The regression coefficients for the linear difference model represented most obviously the actual performance of the engine, whereas calibrated coefficients in the Annex 42 model's 27-term polynomials (Eqs. (4) and (5)) displayed no physical representation of engine performance [24] and a lack of statistical significance in each term. As detailed in [24], a 10-term polynomial was also investigated and was found to have similar coefficient magnitudes to Table 4, with no improvement in the accuracy of the result and even less statistical significance in each term than the linear difference model coefficients.

The p -values for the data regression using Eqs. (24) and (25) are greatly reduced from those achieved using the 27-term polynomial. This means each term has more statistical significance. The very low p -values for the constant terms indicate they are highly significant at the 5% level; this indicates Eqs. (24) and (25) are of a more suitable form than the 27-term polynomials currently used in the Annex 42 model.

Residuals for the higher-order polynomials were lower than for Eqs. (24) and (25); however, the improvement in accuracy at the calibration points does outweigh the potentially erratic behaviour of the higher order model between calibration points.

Plotting the residuals for the calibrated linear difference model against measured predictor variables revealed no clear trend, and so the absence of a regression term cannot be identified specifically by this test.

Table 4
Regression coefficients of linear difference model.

Electrical efficiency			Thermal efficiency		
Regression coefficients	p -Value		Regression coefficients	p -Value	
α_0	1.175exp-01	1.354exp-40	β_0	9.455exp-01	6.736exp-39
α_1	1.825exp-04	3.844exp-07	β_1	4.211exp-04	1.027exp-01
α_2	1.693exp-04	1.789exp-02	β_2	-2.627exp-04	6.826exp-01
α_3	-8.292exp-04	3.098exp-05	β_3	-2.617exp-03	1.009exp-01

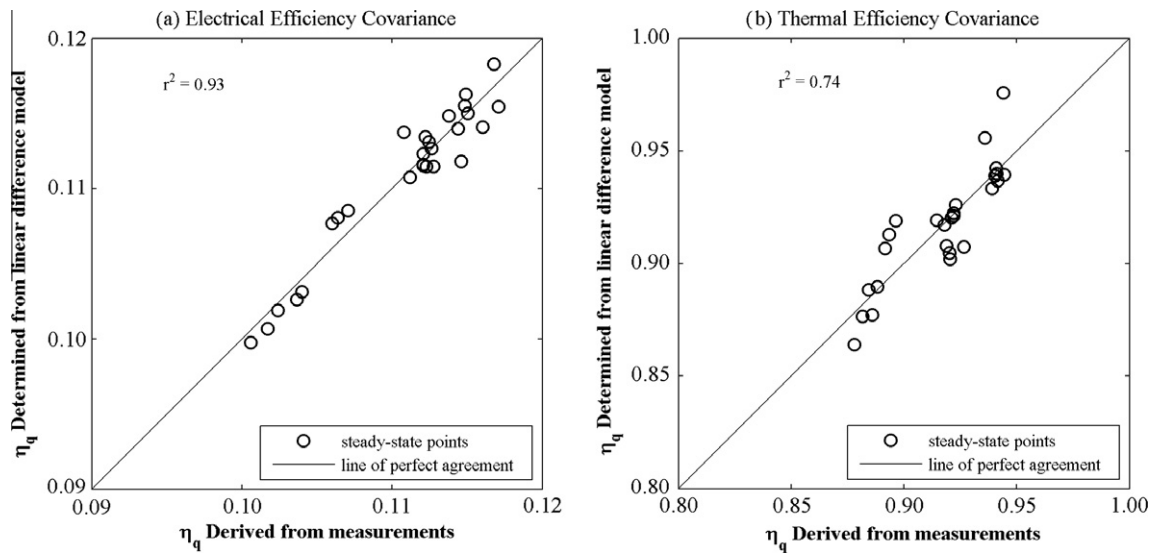


Fig. 3. Coefficient of determination for calibrated linear difference model Eqs. (24) and (25).

Thermal and electrical efficiencies derived from the linear difference model are plotted against values derived from measurements in Fig. 3, showing the basis for calculating the coefficient of determination.

The coefficients of determination for the linear difference model are, for electrical efficiency, 0.93, and for thermal efficiency, 0.74. This indicates the proposed linear difference model accounts for 93% and 74% of the variability in observations of η_e and η_q , respectively. The average percent error of the absolute value of residuals in electrical efficiency is 0.98% and in thermal efficiency is 1.00%. For comparison, the coefficient of determination of the 10-term tri-variate polynomials were 0.89 and 0.96 for electrical and thermal efficiencies, respectively.

Using only one of the above statistical methods does not conclude one model form preferable over any other. However, the overall observation is that the linear difference model holds several advantages over other possible steady-state efficiency models:

- Fewer data are required for calibration.
- Results are less sensitive to skew data or measurement uncertainty.
- Calibration coefficients directly indicate physical engine performance.
- Calibration coefficients could be approximated without any experimentation.
- Each term is statistically significant.
- The linear form produces no oscillatory behaviour between calibration points.

In general, it is recommended that no terms in Eqs. (24) and (25) are excluded from the proposed model at this time for the following reasons:

- The test matrix did not include part-load steady-state testing; a stronger dependency on electrical output is anticipated than that has been shown here.
- The cooling water flow rate is known to affect thermal performance, which in turn affects electrical performance.
- It is possible that other data sets would yield different results for regression coefficients and p -values.

Further work to apply these changes to the existing combustion engine model and validate the results is ongoing.

6. Proposed improvements to modelling start-up and shut-down operation

The methods used in the Annex 42 model to model start-up and shut-down operations were validated using a data set with some critical measurements taken on 10- and 15-min intervals.⁴

6.1. Start-up fuel flow characteristic

The fuel flow characteristics in Eq. (6) were validated by Ferguson et al. [18] using 10-min fuel flow data integrated from one-minute measurements. Fuel flow measurements for a typical start-up period in the current data set are shown in Fig. 4. Measurements from the current data set, recorded instantaneously at 10-s intervals, reveals new information about trends in fuel flow as the engine warms up.

At ignition, the fuel flow for cold starts is higher than the fuel flow for hot starts; however, the trends are identical for all starts:

- Fuel flow steps from zero to a peak value.
- Fuel flow decreases (approximately three minutes for hot starts and four minutes for cold starts for the SE micro-cogeneration device under test).
- Fuel flow increases for 20–30 s beginning when the piston motion and alternator rotation begin (cranking), and
- The fuel flow decreases until steady-state is achieved.

The overall start-up fuel flow trend is similar to the 10-min integrated fuel flow data used in the previous calibration of the model [11], upon which Eq. (7) was based, with one exception being the increase in fuel flow observed in Fig. 4 at about four minutes elapsed time. However, Eq. (7) is a more generic form for modelling fuel flow, and so the fuel flow increases upon alternator cranking considered a trait of the prototype SE micro-cogeneration device under test only, and not SE devices in general. The cumulative fuel consumption over a start-up period is likely represented well by calibrating Eq. (7), and so no modifications to the start-up fuel flow characteristic are recommended. Full calibration of

⁴ Fuel flow data were measured once per minute and average values were recorded every 10 min. Electrical data were measured and recorded every 15 min. Thermal data including temperatures were measured and recorded at 1-min intervals.

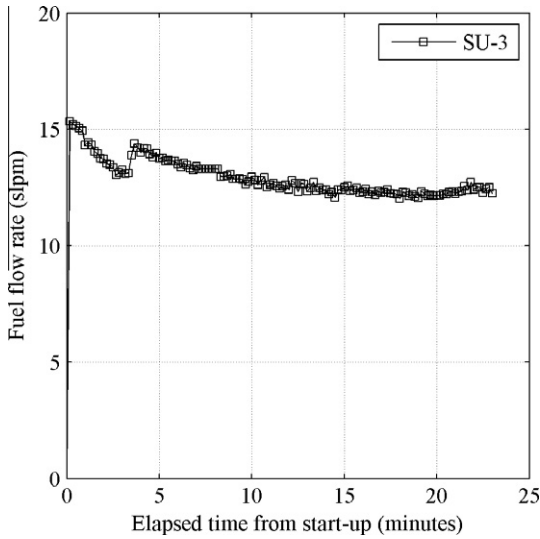


Fig. 4. Start-up fuel flow measurements.

the Annex 42 model using the current data is ongoing and results will be presented in a future publication.

6.2. Start-up electrical output characteristic

The electrical output during start-up is modelled in the Annex 42 combustion cogeneration model by Eq. (8). The validation data set used by [18] for this parameter included only 15-min measurements totalled over the measurement interval. The current data set shows instantaneous behaviour of electrical output at 10-s intervals. A typical graph of net electrical output is shown in Fig. 5 for 30 min of operation after the start-up signal was received by the SE micro-cogeneration device. Detailed measurements depict the following start-up behaviour with respect to net electrical output:

- Constant electrical consumption to operate controls, draft fan, coolant circulation pump, and other auxiliaries.
- Spike electrical consumption to initiate alternator rotation, and
- Positive increasing electrical generation until steady-state operation is achieved.

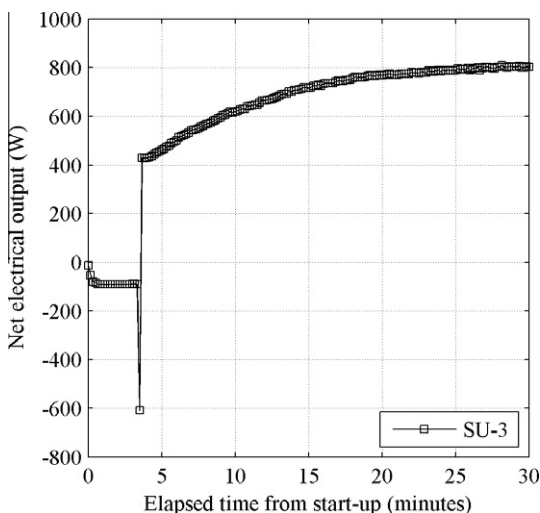


Fig. 5. Start-up net electrical output.

The data measurement interval of 10 s was not sufficient to observe the spike in electrical consumption for cranking during each recorded start-up period. Although the power consumption spike is typical behaviour of a SE for practical purposes such as sizing of auxiliary equipment during an engine installation, it is not important in the overall (cumulative) accuracy of the model, nor on the timescale of building performance simulation. For these reasons, it is recommended that the power consumption spike not be treated in this model.

The end of start-up operation is defined here as the time at which the net electrical output reaches 95% of the maximum value recorded during the first hour of operation. The Annex 42 model defines the end of start-up operation as the time at which the demanded power is produced; however, for the prototype engine tested, the power output is not a value set by the user or by an attempt of the engine to dynamically match the electrical load. In the current data set, it is not practical to use the maximum electrical output as the end of start-up operation since noise may cause the maximum electrical output to occur long after steady conditions have been achieved, and since the cooling water temperature and flow rate were not held constant during start-up.

Two differences between hot and cold starts are identified from the current data set:

- The generator cranks sooner in a hot start, and
- The electricity produced immediately following the generator cranking is greater for a hot start.

The Annex 42 model does include engine temperature in its treatment of start-up electrical output. Calibration and validation of Eq. (8) with the current data set is ongoing; for now, no changes to the form of Eq. (8) are recommended. However, it is recommended that the model be revised to include a period of constant (negative) electrical output before engine cranking.

The recommended alternative to Eq. (8) for start-up electrical output of a SE micro-cogeneration device is given in Eq. (29)

$$P_{net,start-up} = \begin{cases} P_{aux} & \text{for } T_{eng} \leq T_{eng,crank} \\ P_{max} k_p \left(\frac{T_{eng} - T_{room}}{T_{eng,nom} - T_{room}} \right) & \text{for } T_{eng} > T_{eng,crank} \end{cases} \quad (29)$$

where $P_{net,start-up}$ (W) is the net electrical output of the SE micro-cogeneration device during start-up operation, P_{aux} (W) is the electrical generation (negative) of all auxiliary components within the SE micro-cogeneration device, and $T_{eng,crank}$ (°C) is the engine temperature at which generator cranking occurs, calculated from average experimental data.

As with the fuel flow characteristic, the electrical output model will be calibrated and validated and results will be presented in an upcoming publication.

6.3. Shut-down fuel flow characteristic

The shut-down fuel flow characteristic is very simple for the engine under test. The fuel is effectively shut off immediately via a solenoid within the engine enclosure; thus, there is no transient. Therefore, the Annex 42 model's shut-down fuel flow characteristic is endorsed by the current data set, with effectively zero fuel flow throughout the shut-down period.

6.4. Shut-down electrical output characteristic

Previous experimental data provided only a coarse approximation of electrical use during shut-down operation [11]. These data could not provide insight to a model more detailed than estimating

the power generation during shut-down as a constant value, $P_{net,shut-down}$.

Eleven shut-down points were selected from the data collected in this work to represent the engine's behaviour in this mode. Net electrical output measurements from shut-down point SD-6 are shown in Fig. 6. The other data sets are not shown for clarity, but exhibited similar trends. Time zero in Fig. 6 corresponds to the last data point with steady-state fuel flow.

From observing the above (and similar) shut-down cycle data, the shut-down period of a SE cogeneration engine could be characterized by four distinct behaviours in the following sequence:

- From the end of normal operation, the electrical output decreases linearly until a (negative) constant electrical output is reached. This results from the alternator slowing down but still producing some power from residual heat in the engine.
- The electrical output changes linearly to a different (negative) electrical output, representing the alternator stopping.
- The electrical output remains constant at a negative value for a finite period. This represents electrical consumption by auxiliary components including the cooling water pump, draft fan, and electronics.
- The electrical output steps to a (lesser negative) constant value equal to electrical draw by the electronics only; this signifies the end of shut-down operation and the beginning of stand-by operation.

Similar to the treatment of start-up electrical output discussed earlier, the short time it takes for the alternator to stop could be ignored and modelled by a step from the electrical output when the alternator stops to the electrical output when auxiliaries only are operating. This sequence could be described by Eq. (30):

$$P_{net,shut-down} = \begin{cases} k_{SD}\Delta t + P_{net,(t-1)} & \text{while } P_{net,shut-down} > P_{min,shut-down} \\ P_{aux} & \text{for } T_{eng} > T_{eng,cool} \end{cases} \quad (30)$$

where $P_{net,shut-down}$ (W) is the net electrical output of the SE micro-cogeneration device during shut-down operation, k_{SD} ($W s^{-1}$) is a calibrated linear coefficient for the linear reduction in electrical output during shut-down, Δt (s) is the simulation timestep, $P_{net,(t-1)}$ (W) is the net electrical output at the last timestep of operation before shut-down was initiated, $P_{min,shut-down}$ (W) is the observed minimum net electrical output, averaged from all available shut-down segments, and $T_{eng,cool}$ ($^{\circ}C$) is the temperature the aggregate engine

thermal mass must cool to in order for the engine to enter stand-by operation, calculated from average experimental data.

The 10-s measured data from shut-down operation clearly shows that the electricity produced in transient states is not constant and this assumption in the Annex 42 model is not valid. Since a 10-s data collection interval is suggested in the Annex 42 test protocol, and since no extra instrumentation would be required, future data sets collected for Annex 42 model calibration would be sufficient to calibrate a shut-down model that includes Eq. (30).

Further implementation, calibration, and validation of the Annex 42 model including the changes proposed herein are ongoing.

7. Conclusions

IEA/ECBCS Annex 42 produced a zero-order model for simulating the performance of combustion-based micro-cogeneration devices within whole-building simulation programs. This model had been previously calibrated to represent the performance of one particular SE micro-cogeneration device and its validity had been partially assessed using a limited set of empirical data.

The current study critiques the Annex 42 model using an extensive experimental data set that was collected specifically for the purposes of improving this model. These experimental data were gathered from a prototype SE micro-cogeneration device with a net electrical output of 750 W and a thermal output of 5 kW. Measurements of parameters that were critical for model calibration and validation were recorded at 10-s intervals. The test matrix, which was based on a protocol established by Annex 42, included several 30-min-long experiments under steady-state conditions. These experiments spanned a range of cooling water flow rates and engine inlet temperatures in order to produce a data set suitable for critiquing the functional form of the steady-state efficiency performance equations of the Annex 42 model. The new data set also includes start-up and shut-down modes of operation used to assess the Annex 42 model's treatment of transient states.

Based upon the experimental observations, a new "linear difference" formulation is proposed as a replacement to the steady-state efficiency equations in the Annex 42 model. This is found to be more representative of actual engine behaviour, requires fewer experiments to calibrate, provides transparent results, is less sensitive to erroneous calibration data, and provides greater accuracy between calibration points.

The new experimental data have verified the appropriateness of the Annex 42 model's treatment of the fuel flow during shut-down.

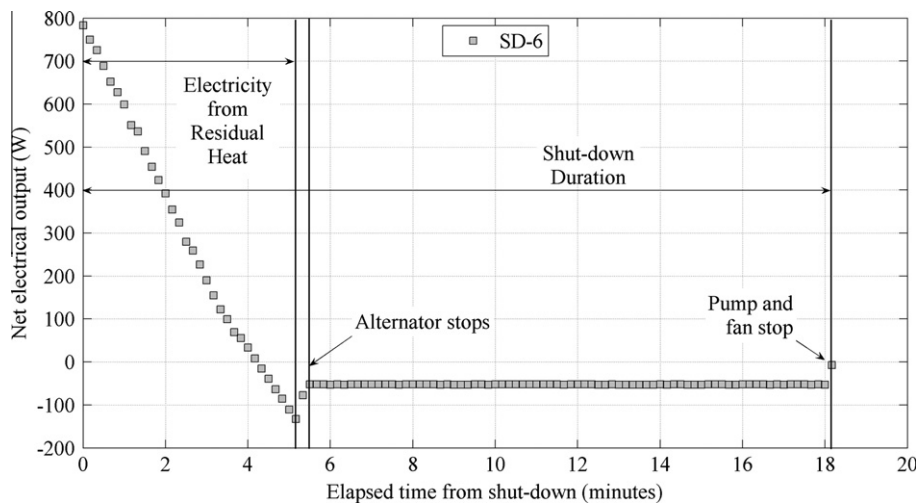


Fig. 6. Shut-down net electrical output.

However, observations based upon these same data have resulted in proposed changes to the treatment of electrical production during both start-up and shut-down. Specifically, new functional formulations are put forward for these transient modes of the Annex 42 model.

It must be noted that these proposed improvements to the Annex 42 model are based upon data gathered from a single prototype SE micro-cogeneration device. The general applicability of the proposed changes should be tested using experimental data from other SE and ICE micro-cogeneration devices.

Future research will incorporate the proposed model changes into a building simulation program. The data set reported here will then be used to calibrate the new model and to validate its implementation into the program. The newly calibrated model could then be used to study the performance of the prototype SE micro-cogeneration device under various operating scenarios, optimize integral components for balance of plant, and examine effects of mass deployment on the electricity grid.

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